

## **Air jet cooling of brake discs.**

*By*

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### **ABSTRACT**

This paper reports on an investigation of a novel approach to the cooling of brake discs, based on the application of impinging air jets. This has the capacity to enhance the heat transfer coefficients at the disc surface quite considerably without affecting the disc design, so that the disc construction may then be optimized without reference to heat transfer. Using a purpose built test-rig, disc temperature histories were recorded using IR thermography for varying jet air flow rates, angle of impingement, dimensionless distance from the brake disc and rotational speed. As well as comparing cooling effectiveness for different test parameters, convective heat transfer coefficients were calculated from the transient temperature data and were used as boundary conditions for a finite element model of the process. The results obtained from this investigation suggest that the higher convection coefficients achieved with jet cooling will not only reduce the maximum temperature in the braking cycle but will reduce thermal gradients, since heat will be removed faster from hotter parts of the disc. Jet cooling should therefore be effective to reduce the risk of hot spot formation and associated disc distortion.

### **1. INTRODUCTION**

Disc brakes are required to transform large amounts of kinetic energy into heat at the contact between the disc and pad. The resulting temperature distribution at the friction

interface can be quite complex, and directly affects braking performance. It has been investigated by several researchers over many years [1–6], who have shown how high temperatures can cause a fall in friction coefficient, or fade, and increased wear of both discs and pads. Both fade and increased wear are affected by local thermal distortions of the disc, which can lead to hot spots [7]. These hot spots are amongst the most dangerous phenomena in brakes, since they are associated with high thermal gradients. The consequent stress fields can lead to low cycle fatigue of the discs, cracking, and even catastrophic failure [8]. Local thermal distortion also concentrates the braking effort over a small fraction of the pad area, which increases local flash temperatures with a resulting increase in wear and fall in friction.

If there is insufficient time for the disc to cool between one brake application and the next, the maximum temperature will rise, exacerbating the above problems. If the overall temperature is high enough, the brake fluid may boil in the caliper cylinders, leading to “fluid fade”, and a potentially dangerous reduction in braking effort. For this reason, brake discs are often designed with slots or holes to create turbulence as they rotate, and thus to enhance convective heat transfer to the air [8,9]. The design constraints on the discs make it difficult to optimize every aspect of their design simultaneously, since it is desirable to optimize the weight and stiffness of the discs as well as their heat capacity and heat transfer capability, which leads to conflicting design requirements.

Impinging jets are known as a method of achieving particularly high heat transfer coefficients and are therefore employed in a broad range of engineering applications. Although liquid jet cooling yields higher heat transfer coefficients, air jet cooling can be very effective and clearly has merits in terms of the simplicity of the cooling system. Thus, El-Sheikh and Garimella [10] investigated the air impingement cooling of heat sinks for electronics, Roger, Buck and Muller-Steinhagen [11] investigated the jet impingement

cooling of a high temperature solar thermal receiver and O'Donovan et al. [12] and Babic et al. [13] used air and mist jet impingement respectively for the cooling of a grinding process.

In order to interpret the results from jet impingement cooling, it is necessary to determine the convective heat transfer coefficients. One common technique for determination of convection coefficients from surface temperature measurements involves a step change in temperature of either the fluid or the test specimen; thus the necessity for electrical surface heating, and the associated geometric constraints, is overcome. Most studies based on this technique have involved the instantaneous introduction of a heated flow field to the specimen [14-16]. However, in the study of Martinez-Botas et al. [17] the test section is heated and then introduced to an ambient temperature flow. In both cases the rate of heat transfer is found by measuring the surface temperature as a function of time. Although most previous investigations have used liquid crystal thermography for these measurements, Ekkad et al. [16] developed an approach based on IR thermal imaging, which determines both heat transfer coefficient and film effectiveness (non-dimensional adiabatic wall temperature) in a film cooling application. Solution of the transient conduction equation with a semi-infinite solid assumption then yields the convective heat transfer coefficient.

In this paper, a novel approach to the cooling of brake discs is investigated. It relies on applying a well-proven method of air cooling – impinging air jets – to the surface of the disc. This has the capacity to enhance the heat transfer coefficients at the disc surface quite considerably, independently of disc design, so that the disc construction may be optimized for strength, weight and stiffness. It should also be possible to achieve much higher heat transfer coefficients than can be attained with current techniques, thereby reducing the thermal problems described above. The main additional requirements are a supply of compressed air onboard the vehicle, and a suitable arrangement of nozzles to deliver it to the discs during braking. In heavy vehicles, such as trucks, coaches and rail vehicles, this should pose no

problem, since compressed air is usually available as standard. For smaller vehicles, though, the expense of a compressor could prove an impediment.

## 2. EXPERIMENTAL TESTING

### 2.1 Experimental Rig and Instrumentation

In order to study the heat transfer from a brake disc in isolation from tribological effects, a special apparatus was designed and built. It is shown in figures 1a and 1b.

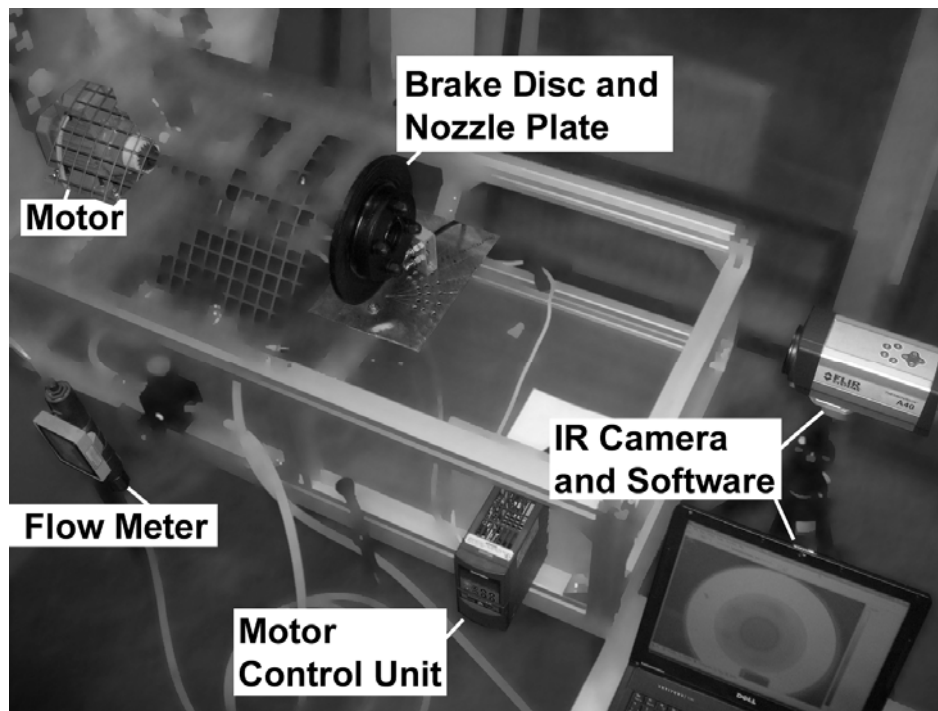


Figure 1a. Experimental apparatus.

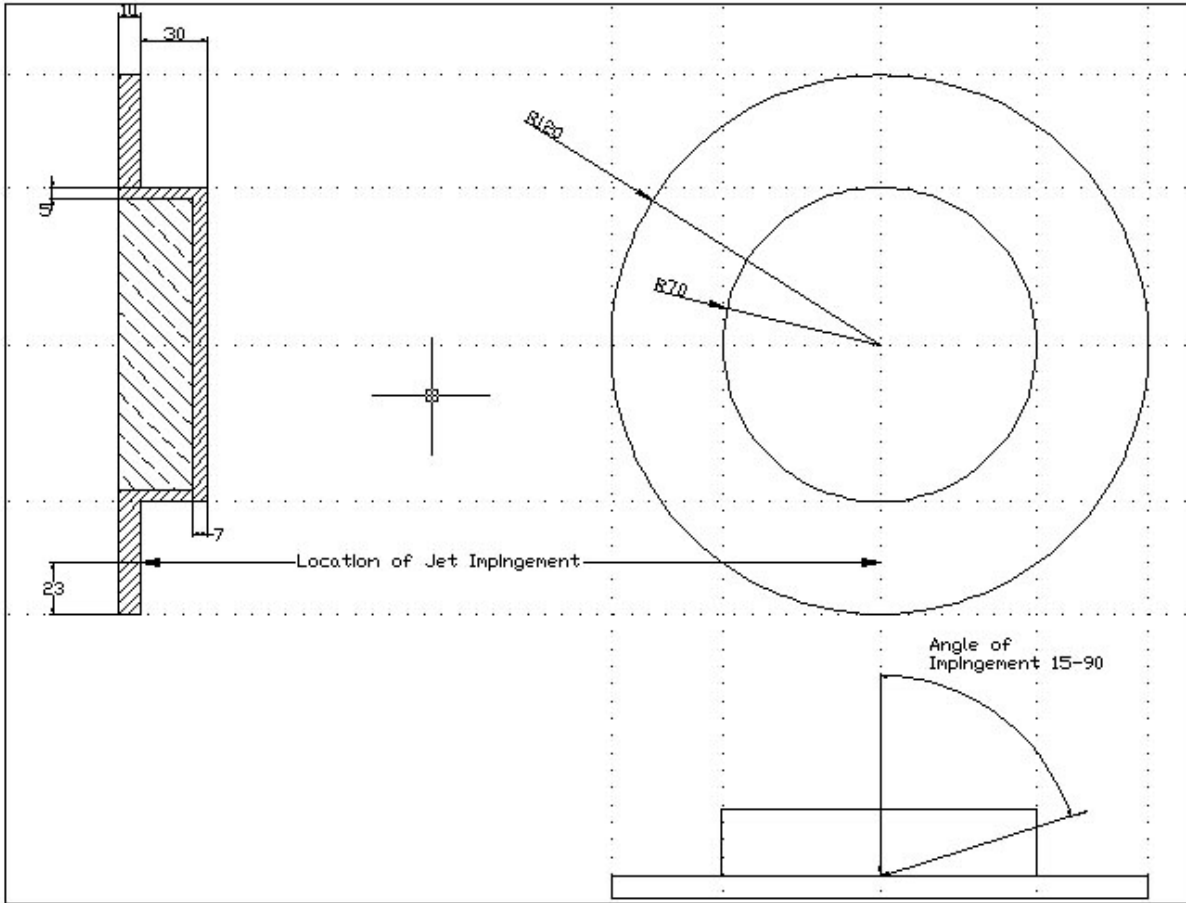


Figure 1b. Schematic of Brake Disc Impingement

A standard automotive cast iron brake disc from a Nissan Micra (flange diameter = 238 mm, hub diameter = 140 mm, flange thickness = 9 mm) was mounted on a shaft, running on two ball bearings, and driven through a coupling by a 3 phase AC motor. The speed of the motor was controlled by a Siemens inverter fixed to the frame of the apparatus. A nozzle plate was mounted on the frame to allow up to four Silvent BS1001 stainless steel Laval nozzles to be directed towards the disc in various orientations. The jets impinged at 23mm from the edge of the brake disc. However, for all the tests reported here, two nozzles were used, one on each side. The nozzles were aligned so as to impinge on a point on the rotor centre line and the fixing method allowed the angle of impingement, defined as the angle subtended between the nozzle and the disc face, and the dimensionless distance from the brake disc,  $H/D$ , to be

varied. These nozzles have a central orifice surrounded by six slots. The effective hydraulic diameter is 5 mm, and the effective throat of the nozzle is downstream of the orifice. Air was supplied to these nozzles from the building's 6 bar compressed air supply; the flow rate was controlled and measured by a Bosch 0821 pressure regulator and a Platon GMT metal tube flow meter respectively. The maximum flow velocity in the jet is supersonic when the supply pressure is over 2 Bar.

IR thermography was used to record the surface temperature distribution on the brake disc using a FLIR ThermoCam A40 thermal imaging camera. The disc was painted with black heat resisting paint with an emissivity of 0.95. This increases the signal to noise ratio, and cuts down reflections, which might otherwise disturb the readings, increasing the accuracy of the temperature measurements. The thermal imaging camera was calibrated against a pre-calibrated K-type thermocouple affixed to the disc and tests were conducted to confirm repeatability and to eliminate the influence of variations in the ambient temperature. The camera records temperatures at 320 by 240 pixels in each image and images were recorded at 5 s intervals for test periods of 250 s. The resolution of the infra red camera is 0.1 °C in the range setting (0 °C to 500 °C) used for these tests and the temperatures reported represent an average across a central (10mm wide) band of the flange. Data transfer was through a firewire 1394 interface to a laptop computer; the ThermoCAM imaging software was used to convert each image to a MATLAB array and further processing took place in MATLAB.

A model brake disc of low thermal conductivity is required to determine convection coefficients by use of the semi-infinite solid solution to the transient conduction equation. This is because, for a given model thickness, the material thermal diffusivity and allowable test times are constrained by the validity of the semi-infinite assumption, Schultz & Jones [18], this details one dimensional conduction. Thus, a Perspex replica of the actual brake disc

has been constructed; this yields an allowable test period of 90 s. This analysis follows the procedure proposed by Ekkad et al. [16], in which the difference between the temperature of the cooling jet and the ambient temperature is accounted for by the use of a cooling effectiveness parameter,  $\eta$ . The heat transfer coefficient,  $h$ , is determined from equation (1), in which  $\alpha$  represents the thermal diffusivity,  $k$  the thermal conductivity and  $erfc$  is the complimentary Gaussian error function; subscripts  $s$ ,  $i$ ,  $c$  and  $\infty$  refer to surface, initial, jet and ambient temperatures respectively.

$$dT = \frac{(T_s - T_0)}{(T_\infty - T_0) - \eta(T_\infty - T_c)} = 1 - \exp\left(\frac{h^2 \alpha t}{k^2}\right) erfc\left(\frac{h\sqrt{\alpha t}}{k}\right) \quad \text{eqn (1)}$$

## 2.2 Test Procedure

In the first series of tests, the flange of the brake disc was heated to 160 °C with a butane torch, whilst being rotated at 100 rev/min. The objective of this process was to heat the brake disc to a high, uniform temperature so that the efficacy of different cooling settings could be investigated. The IR camera was used to monitor the temperature uniformity across the brake disc. Figure 2 shows a typical reading of the disc just after heating had ceased. The temperature varies little circumferentially, but is about 15 °C lower at the edges of the flange than at mid diameter. Following this pre-heating process, the speed of the disc was then increased to a specific test setting (ranging from 500 to 1500 rev/min) for the jet cooling tests. The nozzle to disc spacing was set to 15mm, 30mm or 50mm, corresponding to H/D ratios of 3, 6 and 10 respectively and for each of these settings tests were conducted for angles of impingement of 15°, 45°, 75° and 90°. The total air flow rate, in standard litres/s

through both jets, was set at 4.53, 7.85, 10.96 or 14.2 litres/s. The temperature of the disc as it cooled was monitored with the infra red camera for the test duration of 250 s.

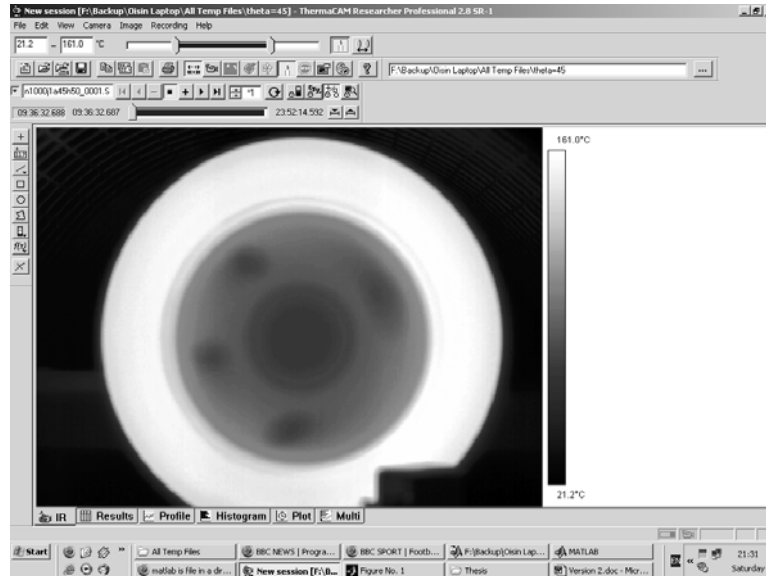


Figure 2. IR camera image of the brake disc at the end of a heating cycle.

A second series of tests was run with the Perspex replica of the brake disc, also painted black. In these tests, the disc was heated with a hot air gun to a uniform temperature of 40 °C. In all tests, jet flow is in the same direction as the rotation of the disk. Initial tests had shown this to obtain higher cooling rates than contra flow. Uniformity of the surface temperature was determined from thermal imaging whereas heat penetration within the Perspex was verified by a fine gauge thermocouple located within the flange. Once a stable and uniform temperature was established, cooling tests were conducted as before. The objective here was to obtain transient surface temperature data for determination of the convective heat transfer coefficients, as described in section 2.1.

### 3. RESULTS



### 3.1 Cooling curves for the brake disc

#### Baseline tests without jet cooling

The first set of tests investigated the effect of disc rotational speed on the cooling curves. No jets were used so as to provide a baseline from which subsequent results could be assessed. Six different speeds were used, giving the curves shown in figure 3. The effectiveness of cooling can be assessed by the time taken for the surface of the disc to lose half its initial temperature excess ( $t_{1/2}$ ), i.e. the time corresponding to  $T_s - T_\infty = 0.5(T_{initial} - T_\infty)$ . At 500 rev/min, it takes 341s for this level of cooling to take place, whilst at 1500 rev/min the corresponding time is 193s. For this configuration, convective cooling results only from the relative motion between the rotating disc and the stationary air; thus disc speed has a significant effect on cooling time. However, when these tests were repeated with one of the least effective jet cooling configurations (impingement angle of  $15^\circ$ , dimensionless disc to jet spacing,  $H/D$ , of 10 and flow rate of 7.85 l/s), the cooling curves at different disc speeds showed much less variation, giving  $t_{1/2} = 108$ s at 500 rev/min and  $t_{1/2} = 87$ s at 1500 rev/min. As the effect of disc speed on cooling was small compared to that of the jet, a speed of 1500 rev/min was used in all subsequent tests.

#### Jet cooling tests

In the next series of tests, the effects of varying flow rate and  $H/D$  were investigated for an impingement angle of  $90^\circ$  and a disc rotational speed of 1500 rev/min. The results are summarised in figure 4. For comparison, without the jets,  $t_{1/2} = 193$ s. At the highest flow rate (14.2 l/s), the jets cool the disc 3.3 times faster than it cools without jets. What is just as interesting is that moving the jets closer to the disc achieves nearly as much cooling at the lowest air flow rate: with  $H/D = 3$ , and a flow rate of 4.53 l/s, the cooling is still 2.6 times

faster than with no jets. This could have important implications for the design of practical cooling systems for brakes where a special supply of compressed air is required.

A third series of tests was run to investigate the effect of impingement angle on the cooling, as discussed by [19]. Figure 5 shows how the value of  $t_{1/2}$  changes with flow rate and angle, for a dimensionless disc to jet spacing of  $H/D = 10$ . Two points may merit further investigation: firstly, the minimum evident in  $t_{1/2}$  at  $75^\circ$  for the two lower flow rates, indicative of an optimal cooling angle. This is broadly similar to the angle obtained by Roy and Patel [20]. Secondly, that at an impingement angle of  $15^\circ$ ,  $t_{1/2}$  seems higher, and cooling less effective, as the flow rate rises to its maximum. Once again, it can be seen how, by carefully selecting the design parameters, cooling can be optimized for low flow rates, so that results nearly as good as those at much higher flow rates can be achieved.

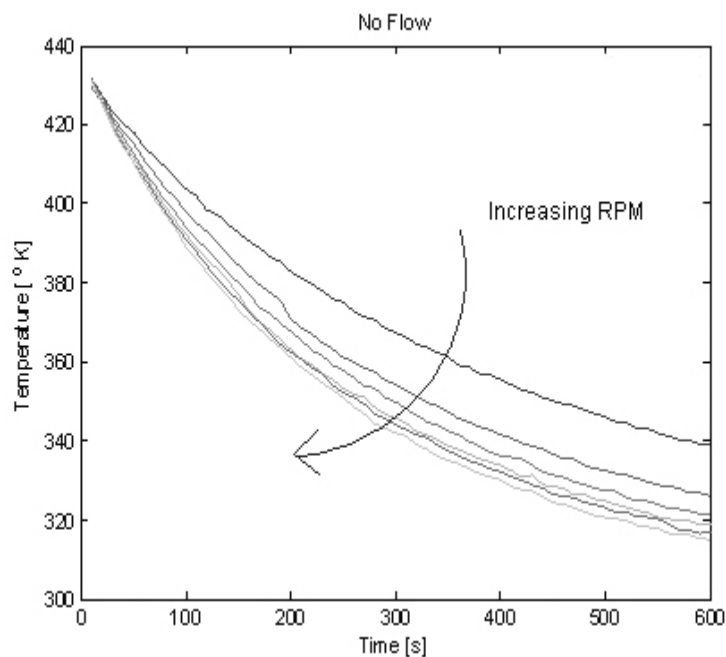


Figure 3. Cooling curves with no jets: the effect of rotational speed.

Two more series of tests were run at  $H/D = 6$  and  $H/D = 3$  to find how impingement angle influenced cooling when the jet was set closer to the disc. Both series were conducted

at the lowest flow rate (4.53 l/s), and both showed that when the jet was moved closer to the disc, impingement angle had very little effect on the cooling. Thus impingement angle only seems to be a significant design consideration if the jet is far enough away from the disc.

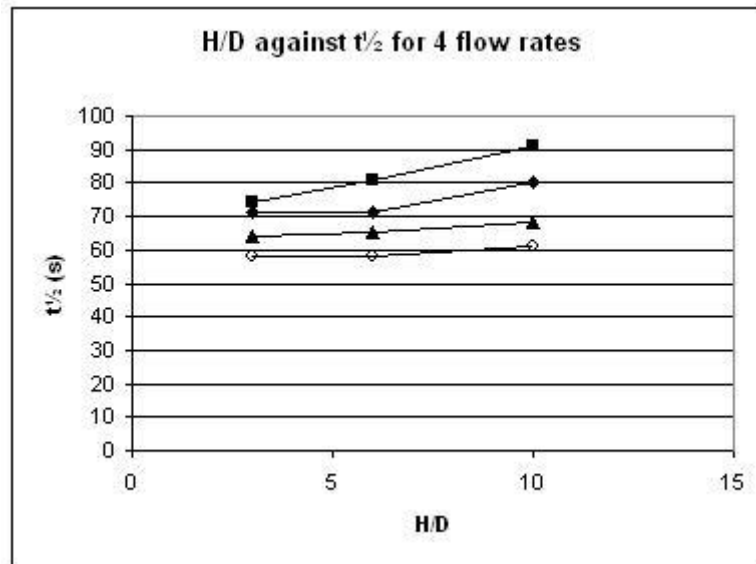


Figure 4. Effectiveness of cooling ( $t_{1/2}$ ) as a function of  $H/D$  and flow rate.

■ 4.53 l/s    ◆ 7.85 l/s    ▲ 10.96 l/s    ◇ 14.2 l/s

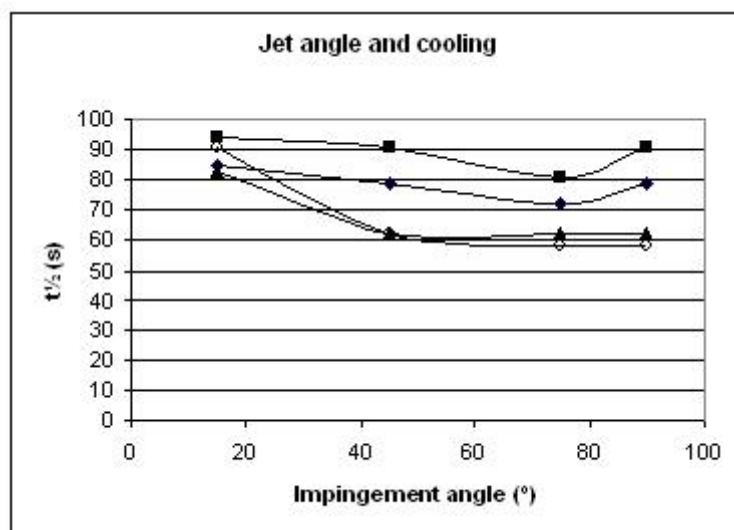


Figure 5. Effectiveness of cooling ( $t_{1/2}$ ) as a function of impingement angle and flow rate ( $H/D = 10$ ).

■ 4.53 l/s    ♦ 7.85 l/s    ▲ 10.96 l/s    ◇ 14.2 l/s

### 3.2 Inference of heat transfer coefficients from transient conduction analysis

The results from the cooling of the Perspex brake disc were analysed using the transient conduction analysis described in section 2.1, in a manner similar to [16]. This yields local and spatially averaged heat transfer coefficients for the full range of test parameters, including tests with no jet cooling but variable disc spin speeds. These coefficients are compared with the actual cooling trends reported in the previous section. In addition, the experimentally determined heat transfer coefficients act as boundary conditions for a finite element model of the jet cooled brake disc; this is discussed further in section 4.

For the baseline tests, it was found that the heat transfer coefficients with no applied cooling were substantially lower than the jet cooled coefficients, and varied as expected with disc spin speed. For example, for  $H/D$  of 10, angle of  $90^\circ$  and spin speed of 1500 rpm the heat transfer coefficient is less than one third of that recorded for the lowest jet flow rate in the same configuration.

From Figure 6 for the average heat transfer coefficients, it can be seen that the effects of air flow rate and dimensionless disc to jet spacing are consistent with the measured cooling times reported in the previous section. Thus, the average heat transfer coefficient rises with decreasing  $H/D$  ratio, which is consistent with research by Baydar et al [21]; it was found also to increase with rising air flow rate as anticipated by results of Martin [22]. The trend with angle of impingement is not quite as clearcut; the heat transfer coefficient does increase initially with increasing angle, which is consistent with the trends shown in Figure 4, but the

heat transfer coefficient appears to reach a plateau rather than peak at 75°. (Geometrical constraints prevented testing at 15° for  $H/D$  of 3.) Although the difference in trend with varying impingement angle may warrant further investigation, the discrepancy is not significant, in the context of the measurement set-up.

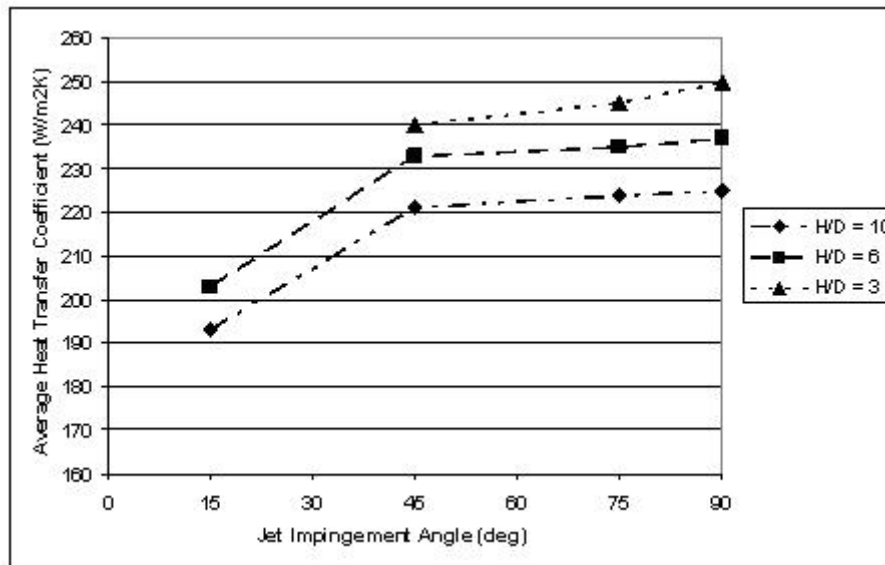


Figure 6 Variation in heat transfer coefficients with  $H/D$  and Impingement Angle, (7.85 l/s)

Although average heat transfer coefficients are of practical relevance, local coefficients over the disc flange may provide additional insight into the convective cooling process. Figure 7 shows the radial variation in heat transfer coefficient, superimposed on the actual brake disc, for jet impingement at  $H/D$  of 10, flow rate of 7.85 l/s and angle of 90°. Figure 8 shows a more conventional representation of local heat transfer coefficients for a flow rate of 7.85 l/s and an angle of 90°, for  $H/D$  values of 3, 6 and 10. Two key features stand out. Firstly, for the lowest disc to nozzle spacing,  $H/D=3$ , secondary peaks in the local heat transfer distribution

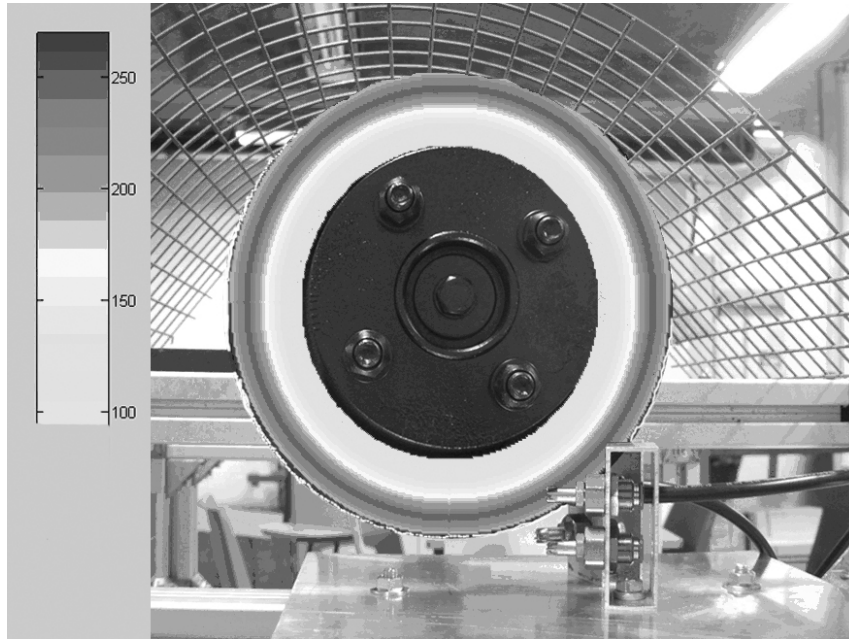


Figure 7 Variation in heat transfer coefficient with radius, superimposed on actual rig

are evident at radial locations of around  $r/D = 1$  and 2. In studies by Baughn and Shimizu [23], Huang and El-Genk [24], O'Donovan and Murray [25] and others, secondary peaks in the heat transfer distribution from a stationary surface to an impinging air jet have been reported; in some cases two radial peaks are present in the heat transfer distributions. These peaks have been attributed, in general, to an abrupt increase in turbulence in the wall jet boundary layer [25]. Secondly, in all cases the heat transfer coefficients are highest near the outside edge of the disc. In jet impingement on a stationary, flat surface, normal impingement would be expected to yield symmetry about the jet centre axis. In the current situation, the asymmetry in heat transfer evident in Figure 8 must be due, in large part, to the flow field associated with the spinning brake disc, which would yield higher velocities at the outer edge of the flange.

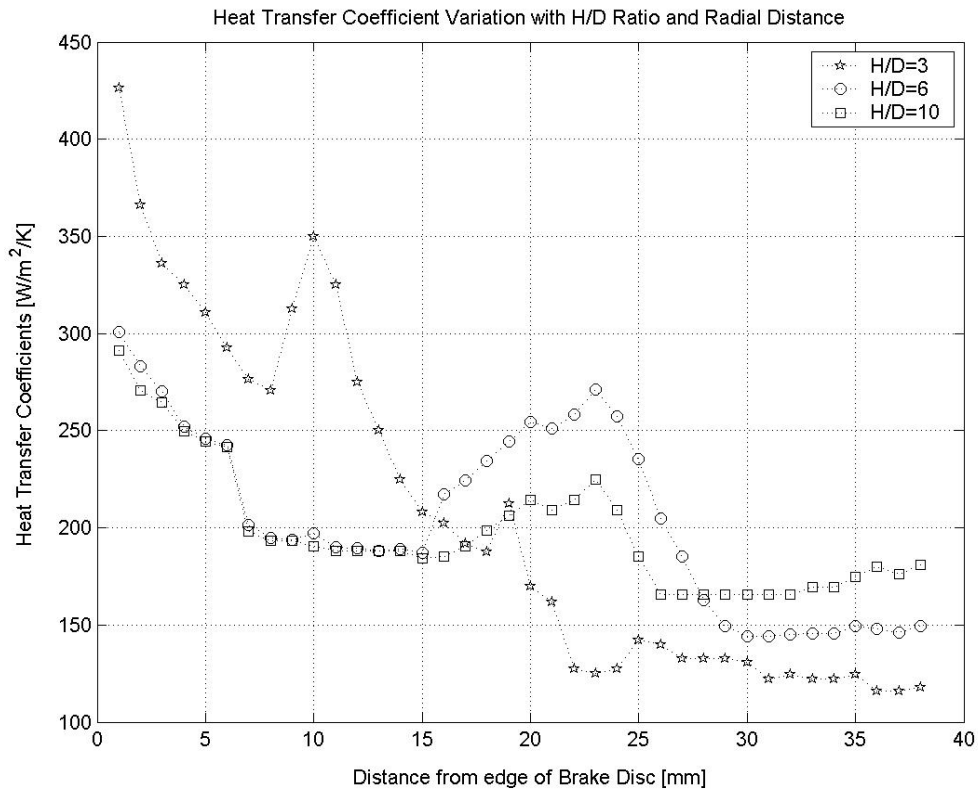


Figure 8 Radial variation in heat transfer coefficient, at different H/D ratios

#### 4. FINITE ELEMENT MODELLING

In order to study the likely response of a brake disc to jet cooling after an emergency stop, a finite element model of the disc was developed in ANSYS ®. The experimentally determined heat transfer coefficients are applied as boundary conditions in the model. The material properties used for the disc, taken from reference [26], are given in table 1. The element used is the 3-D thermal solid SOLID70, a commonly used element in thermal modeling. A mesh was created and refined using the mesh refine tool until a sufficient degree of accuracy was reached.

Thermal conductivity $\text{Wm}^{-1}\text{K}^{-1}$	57
Density $\text{kg/m}^3$	7100
Specific heat $\text{J.kg}^{-1}\text{K}^{-1}$	452
Coeff. of thermal expansion $\text{K}^{-1}$	$1.1 \times 10^{-5}$

Table 1. Properties of cast iron

The model was first used to predict the cooling curve of a trial test, where the brake disc was heated to a temperature of 200 °C and allowed to cool at 1500 rev/min with no jets. The predicted and measured cooling curves are shown in figure 9. It can be seen that the correspondence between the model and the experimental cooling curve is good, providing confidence in the finite element model.

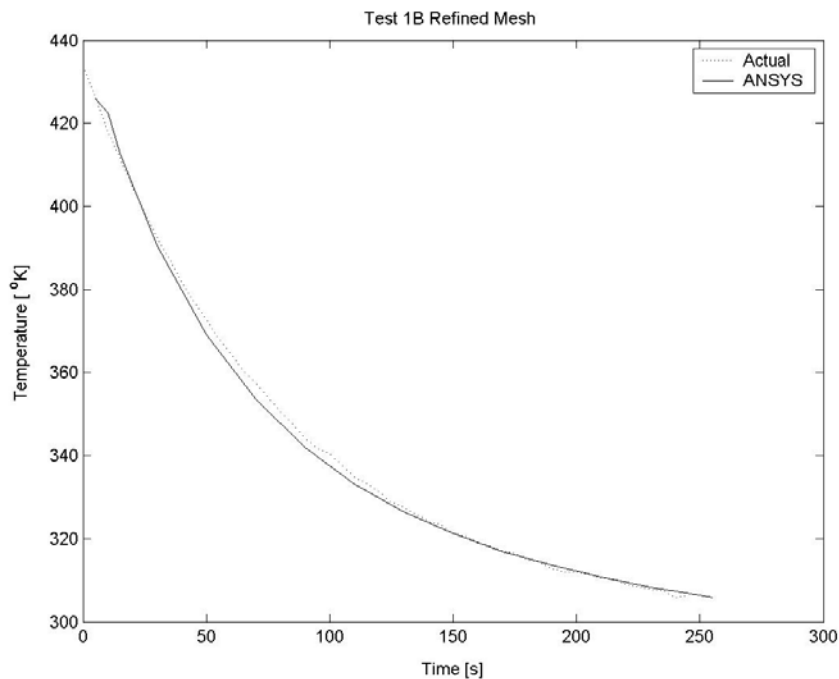


Figure 9 Predicted and measured cooling curves from 200 °C



In an emergency stop when most of the kinetic energy of the vehicle is transmitted to the discs as heat, the disc may reach a temperature of 500 °C [27, 28]. Figure 10 shows the predicted cooling curves without jets, and with jet cooling at  $H/D = 3$ , flow rate = 7.85 l/s and impingement angle of 90°. The measured cooling curves from 200 °C, which were shown to mimic that of 160 °C, are superimposed on each predicted curve to show its accuracy. The correspondence between measured and predicted values at lower temperatures is good, which lends more confidence to the higher temperature predictions. It is clear that jet cooling offers scope for removing heat from the brake disc much more rapidly than just the flow induced by the spinning disc, and may thus provide a means of alleviating some of the thermal problems encountered in practical situations.

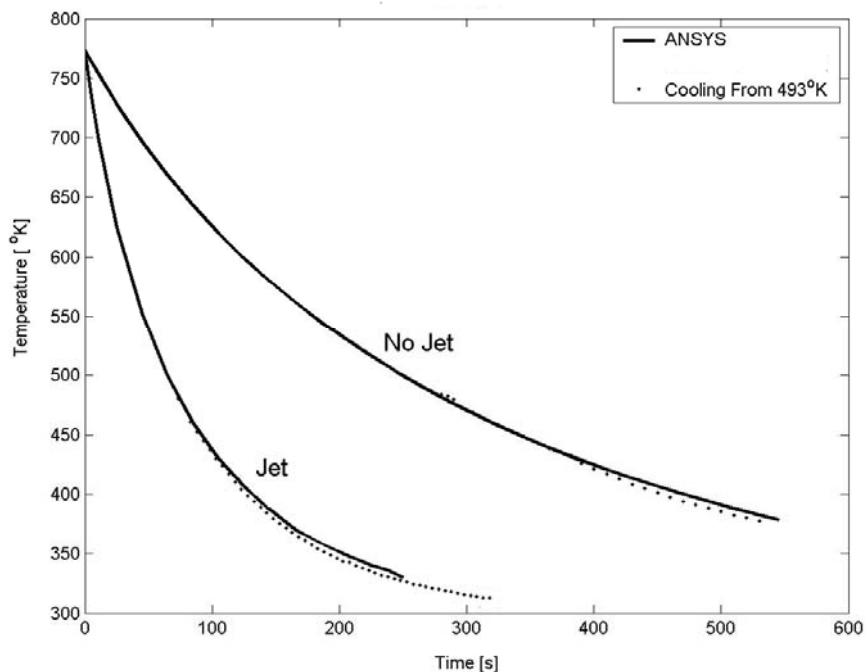


Figure 10 Predicted cooling curves from 500 °C, with and without jets, with measured results superimposed.

## 5. DISCUSSION.

One of the most dangerous conditions in frictional brakes is the formation of macroscopic hot spots (MHS). These occur when the thermal gradients produced in the disc produce stresses high enough to cause yielding. The disc then buckles circumferentially so that all the braking effort is put into the high spots, which then get hotter and hotter [7]. The high convection coefficients which can be achieved with jets may offer a way of alleviating this problem by reducing the thermal gradients in the disc, which are responsible for thermal stress.

We have already shown that the rate of cooling of a brake disc after the application of the brakes is much faster with jet cooling than with natural convection. However, it would also be interesting to see how a higher convection coefficient might influence the heating of the disc during braking itself. The results of the simulated railway braking tests reported by Panier et al. [7], shown in table 2, give almost enough information to allow this to be done, but a few extra assumptions must be made. First, it will be assumed that the maximum braking power is applied continuously during the stop, though clearly it will fall when the vehicle slows sufficiently. Secondly, the material properties of the disc will be assumed to be the same as those of the disc used in our work, and that the thickness of the disc is 80 mm. To obtain an analytical solution of the problem, it will be further assumed that the disc heats uniformly through its thickness when the brakes are applied, an assumption which will tend to underestimate the effect of convective cooling. In reality, the surface temperature will be higher than the bulk temperature of the disc, leading to increased convection for a given mean disc temperature.

Test no.	Speed (km/hr)	Disc OD (m)	Disc ID (m)	Braking power (kW)	T max (°C)	$\Delta T$ (K)	Mass (Kg)	Stopping time (s)	Energy (MJ)
<b>1</b>	<b>60</b>	<b>0.64</b>	<b>0.38</b>	<b>56</b>	<b>269</b>	<b>244</b>	<b>30000</b>	<b>403</b>	<b>15</b>
<b>2</b>	<b>60</b>	<b>0.64</b>	<b>0.38</b>	<b>56</b>	<b>211</b>	<b>186</b>	<b>30000</b>	<b>235</b>	<b>6.4</b>

<b>3</b>	<b>90</b>	<b>0.64</b>	<b>0.38</b>	<b>84</b>	<b>214</b>	<b>189</b>	<b>30000</b>	<b>263</b>	<b>9.4</b>
4	120	0.64	0.38	164	300	275	27000	193	15
5	200	0.64	0.38	150	300	275	9800	215	15

Table 2. Braking test results with estimates of the potential effect of jet cooling.

[The results in bold are for brake applications which initiated MHS

The figures in italic are calculated from the results in this paper]

$\Delta T$  is calculated assuming an ambient temperature of 25 °C

With these assumptions, it is possible to write the following differential equation for the temperature of the disc,  $\theta$ , as a function of the braking time,  $t$ , and the initial temperature,  $\theta_0$ .

$$\frac{d\theta}{dt} = \frac{Q - hA(\theta - \theta_0)}{MC} \quad \text{eqn (2)}$$

where  $Q$  is the braking power,  $h$  the heat transfer coefficient,  $A$  the area of the heat transfer surface,  $M$  the mass of the disc, and  $C$  its specific heat. With a little manipulation, this yields the following solution for the temperature after a braking time  $t$ .

$$\theta - \theta_0 = \frac{Q}{Ah} \left( 1 - e^{-\frac{Ah}{MC}t} \right) \quad \text{eqn (3)}$$

Using the figures given in table 2 for the first test, we get the predicted heating curves shown in figure 11. The upper curve assumes  $h = 60 \text{ W/m}^2/\text{K}$ , representing the naturally cooled case, whilst the lower curve assumes  $h = 300 \text{ W/m}^2/\text{K}$ , representing applied jet cooling. Ambient temperature is assumed to be 25 °C. Where the upper curve reaches the maximum temperature rise measured in the test, the lower curve, representing jet cooling, lies some 50 °C lower. Smaller reductions were predicted with the other tests, since they are of shorter

duration, leaving less time for convection to dissipate the heat. Table 3 shows the reduction in temperature predicted for each test at the moment where the maximum recorded temperature is reached.

Test no.	1	2	3	4	5
Predicted fall in temperature ( $^{\circ}\text{C}$ )	48.8	29.7	19	21.4	23.9

Table 3 Predicted reductions in temperature at the maximum with jet cooling

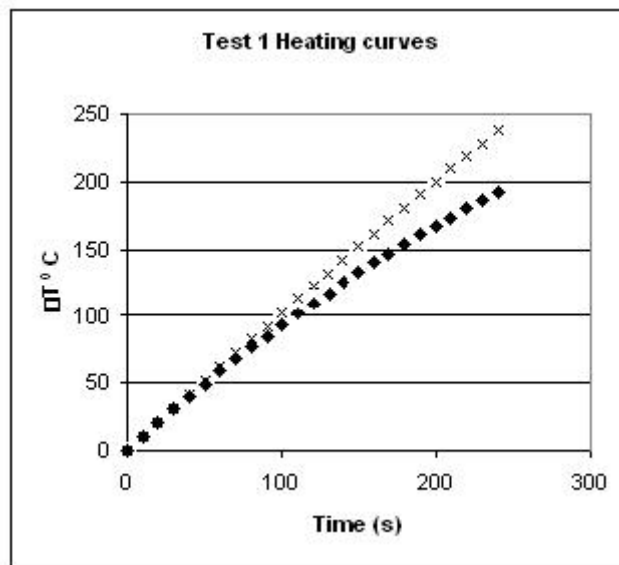


Figure 11 Predicted heating curves for test number 1. The upper curve assumes natural cooling, and the lower one, applied jet cooling.

The higher convection coefficients achieved with jet cooling will not only reduce the maximum temperature, but will naturally tend to reduce thermal gradients, since heat will be removed faster from hotter parts of the disc. Jet cooling should therefore be very effective in two ways to reduce the risk of MHS formation. A still more powerful effect could be

achieved by directing the jets so that the local heat transfer coefficients are higher where the hot spots form. This would probably require a control system, since the radius at which hot bands, the precursors of MHS, form varies with time [7].

Another question of interest is the temperature attained by an automotive brake disc during an emergency stop. In this case, if uniform deceleration,  $a$ , is assumed from an initial velocity  $V_0$ , the differential equation for temperature is modified as follows:

$$Q = FV = F(V_0 - at) \quad \text{eqn (4)}$$

$$\frac{d\theta}{dt} = \frac{F(V_0 - at) - hA(\theta - \theta_0)}{MC} \quad \text{eqn (5)}$$

This equation was solved numerically, piecewise, with a time step of 57 ms for a vehicle equipped with the brake discs studied in this work. It was assumed that the effective mass braked by each front disc was 500 kg, and that the vehicle decelerated uniformly at  $7.848 \text{ m/s}^2$  (0.8g) from 160 km/hr to rest. Three values of heat transfer coefficient were used,  $50 \text{ W/m}^2/\text{K}$ , corresponding to natural cooling,  $250 \text{ W/m}^2/\text{K}$ , and  $400 \text{ W/m}^2/\text{K}$ ; the results are shown in figure 12. The relative reduction in temperature is less than in the longer-lasting railway braking tests, since there is much less time available for convection to remove heat. There could still be the potential for avoiding hot spots, but the advantages are clearly more limited. The maximum temperatures predicted for each heat transfer coefficient are shown in table 4.

h	50	250	400
$\theta_{\text{max}}$	561	550	543

Table 4. Maximum predicted temperatures during an emergency stop for various H

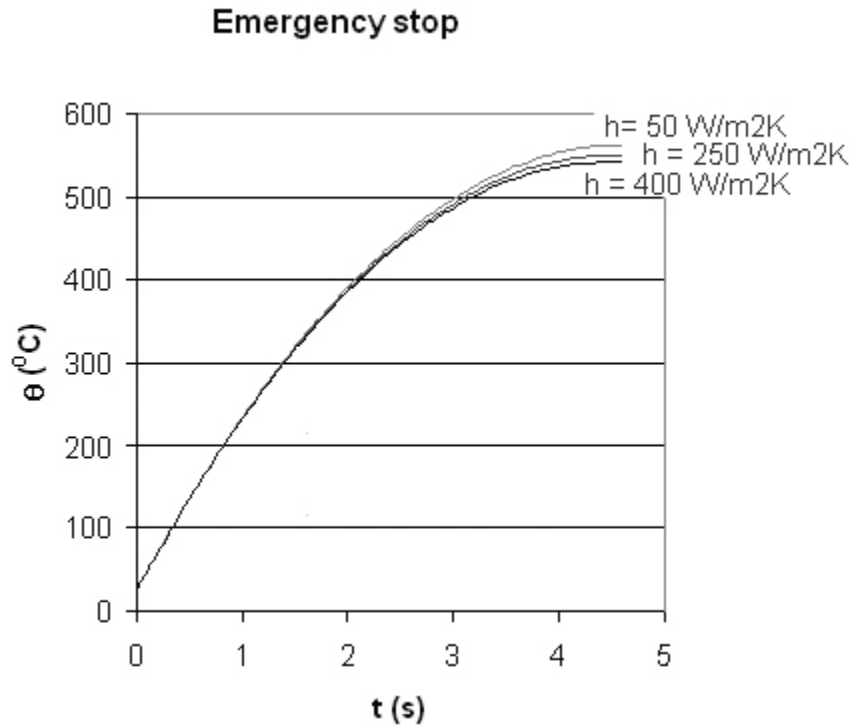


Figure 12 Temperatures predicted during an emergency stop from 160 km/hr

It would, of course, be possible to enhance convection further by injecting a water mist into the air jet, as we have shown elsewhere [13]. However, there is experimental evidence [29] that this would reduce the coefficient of friction between the brake pads and the disc, leading to a different set of problems.

## CONCLUSIONS

A novel approach to the cooling of brake discs, based on the application of impinging air jets, has been investigated. A test-rig has been designed and constructed to allow brake disc temperature histories to be recorded using IR thermography for varying jet air flow rates, angle of impingement, dimensionless distance from the brake disc and rotational speed. The experimental data obtained have been used to compare the cooling effectiveness of different jet configurations. In addition, the disc temperature histories have been used to estimate convective heat transfer coefficients, using transient conduction analysis; these coefficients

were subsequently applied as boundary conditions for a finite element model of the process. The results obtained from this investigation suggest that there are benefits to be gained in the long braking cycles used on heavy vehicles. The higher convection coefficients achieved with jet cooling will not only reduce the maximum temperature in the braking cycle but will reduce thermal gradients, since heat will be removed faster from hotter parts of the disc. Jet cooling should therefore be effective to reduce the risk of hot spot formation and associated disc distortion. Implementation would simply require plumbing the nozzles into the vehicle's compressed air supply. For light vehicles, the benefits are less, and implementation is more problematic. Although maximum temperatures may be reduced slightly during short, emergency brake applications, the main benefit of jet cooling in this case will be to restore the disc more rapidly to ambient temperature.

More research is required to optimize the cooling process and determine the engineering cost, in terms of energy and life cycle cost, so as to evaluate it with respect to other thermomechanical options.

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## APPENDICES

### 1. NOMENCLATURE

a	Acceleration
A	Area [m <sup>2</sup> ]

F	Braking Force
Q	Braking Power [J]
x, y, z	Cartesian Coordinate Variables
$\eta$	Cooling Effectiveness
D	Hydraulic diameter [m]
h	Local Heat Transfer Coefficient [ $\text{W}/\text{m}^2/\text{K}$ ]
M	Mass [kg]
C	Specific Heat Capacity [J/kg/K]
r	Radius [m]
T, $\theta$	Temperature [ $^{\circ}\text{C}$ ] or [K]
k	Thermal Conductivity [ $\text{W}/\text{m}/\text{K}$ ]
$\alpha$	Thermal Diffusivity [ $\text{m}^2/\text{s}$ ]
t	Time [s]
$t_{1/2}$	Time taken for the surface of the disc to lose half its initial temperature excess [s]
V	Velocity
Subscripts	
0	Initial
s	Surface
c	Coolant
$\infty$	Ambient